NPS69-86-012PR

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DYNAMIC MODELLING OF AN ELECTROMECHANICAL VALVE USING FREQUENCY RESPONSE DATA

by

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REPORT DOCUMENTATION PAGE

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2a. SECURITY CLASSIFICATION AUTHORITY	3. DISTRIBUTION/AVAILABILITY OF REPORT Approved for public release; distribution is unlimited.
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE	
4. PERFORMING ORGANIZATION REPORT NUMBER(S) NPS69-86-012PR	5. MONITORING ORGANIZATION REPORT NUMBER(S)
6a. NAME OF PERFORMING ORGANIZATION Naval Postgraduate School 6b. OFFICE SYMBOL (If applicable)	7a. NAME OF MONITORING ORGANIZATION
6c. ADDRESS (City, State, and ZIP Code) Monterey, CA 93943	7b. ADDRESS (City, State, and ZIP Code)
8a. NAME OF FUNDING/SPONSORING ORGANIZATION 8b. OFFICE SYMBOL (If applicable)	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER
8c. ADDRESS (City, State, and ZIP Code)	10. SOURCE OF FUNDING NUMBERS PROGRAM ELEMENT NO. / PROJECT NO. / TASK NO. / WORK UNIT ACCESSION NO.

11. TITLE (Include Security Classification)
Dynamic Modeling of an Electromechanical Valve Using Frequency Response Data

12. PERSONAL AUTHOR(S)
R. L. Miller, D. L. Smith

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Progress	FROM 6/86 TO 9/86	86/12/15	16

16. SUPPLEMENTARY NOTATION

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Control, Dynamic Modelling

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

Dynamic modelling of electromechanical parts is often accomplished by conducting an analysis from first physical principles. While this is always a useful effort for physical insight, it may lead to transfer function models which are unnecessarily complex. This paper presents a supplementary dynamic modelling method based on frequency response data. The method is applied to modelling of a globe valve use for water flow control.

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT ☒ UNCLASSIFIED/UNLIMITED ☐ SAME AS RPT ☐ DTIC USERS	21. ABSTRACT SECURITY CLASSIFICATION Unclassified
22a. NAME OF RESPONSIBLE INDIVIDUAL David L. Smith	22b. TELEPHONE (Include Area Code) (408) 646-3383 22c. OFFICE SYMBOL 69Sm

DD FORM 1473, 84 MAR — 83 APR edition may be used until exhausted. All other editions are obsolete

SECURITY CLASSIFICATION OF THIS PAGE

Unclassified

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I. Previous Work

Frequency data analysis of mechanical components has either not been often used or not often published. In any case, the literature has examples of this type of analysis applied to a stirling engine (1), a boiler (2), friction effects (3), a relief valve (4), and an electrohydraulic servovalve (5). At the Naval Postgraduate School we have used this type of analysis on a diesel engine (6) and a flow control globe valve (7). The method has worked very well and consistently for us and we are planning to apply it to various other mechanical components in the near future. This paper presents the approach, results, and lessons learned in the globe valve study mentioned above.

The system of interest was a small gas generator coupled to a Clayton wate Dynamometer, Model 17-300-CE. The dynamometer was capable of providing an 85% load change in fifteen seconds through a load control system designed and implemented by P. N. Johnson (8). The water load and unload valves were 1 inch globe valves which were actuated by 72 rpm synchronous motors. The load valve inlet pressure was maintained with a 40 psig pressure regulator. The unload valve supply pressure was fixed by pressurizing the dynamometer shell to 4 psig. This was done to assist in the unloading of the dynamometer to meet the design specifications set forth by Johnson.

As a preliminary excersize, the valves were first modelled in the steady-state. Based on flow measurements, an equation for the steady flow as a function of input voltage to the valve actuators was formulated. The resulting cubic equation for the load valve flow rate was

$$Q_{out} = 1.1634V_i^3 - 10.9419V_i^2 + 53.2527V_i - 7.8866,$$

Q in pounds per minute,

V_i in volts.

The results of using these models for the load and unload valves (e.g. ignoring valve dynamics) can be graphically seen in the system response curves shown in figures 1 and 2. The curves show that while the load valve model accurately reflects the actual valve performance, the unload valve model is in significant error. The model formulation used by Johnson for these identical valves was somehow inadequate due to the differing fluid mechanical behavior. In this way it was found that a move substantial modeling technique was necessary for the unload valve. However, since the valve motor, fluid dynamics, and electronics were very complicated, a frequency modelling technique was selected to identify significant dynamic performance.

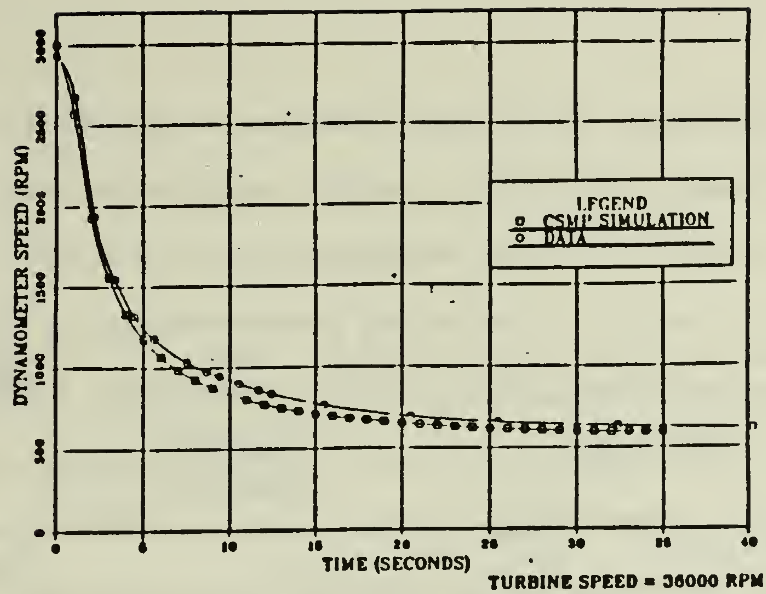

Figure 1. Dynamometer Loading Response

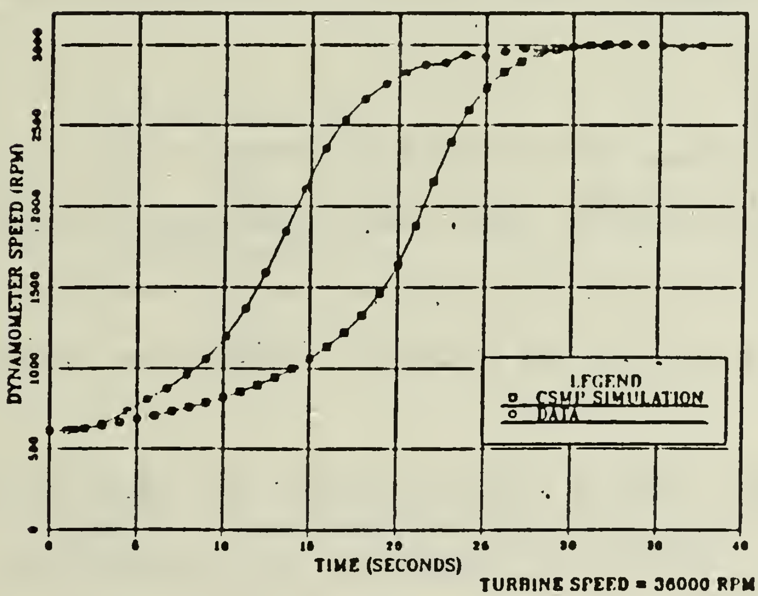

Figure 2. Dynamometer Unloading Response

II. Approach

In order to determine the transfer function of the unload valve for steady state and dynamic conditions, specific experiments were devised and conducted by J. E. Roger (7). The experimental apparatus used was such that water was supplied directly to the unload valve by bypassing the dynamometer. Figure 3 shows the functional setup of the test apparatus and the instrumentation for the collection of data. Roger initially recorded data on flow rate, Q, versus input voltage V_i, for various upstream pressures, P_u, as shown in figure 4. The figure shows a plot of valve position (1 Div = 0.4 volts) and it shows the effect of deadband in the valve electronics. The valve actually began to open at about 0.3v, at the time the input voltage reached 0.4v (1 Div) the valve was only slightly open.

An upstream pressure of 4 psig was then chosen for the formulation of the steady state relationship between Q and V_i. The resulting second order polynomial was

$$Q_{out} = -4.1704V_i^2 + 42.049V_i - 13.6472.$$

An inspection of this equation shows that if the input voltage, V_i, is zero (valve closed) the flow rate, Q, is not zero as it should be. This means that the simulation must zero out the flowrate if the voltage is less than the opening voltage.

In order to get a useful transfer function the dynamic relationship between flow rate, Q, and input voltage, V_i, must be determined. If a constant upstream pressure, P_u, is assumed then the following equation applies.

$$\left.\frac{dQ}{dV_i}\right)_{P_u = c} \triangleq \frac{q}{v}$$

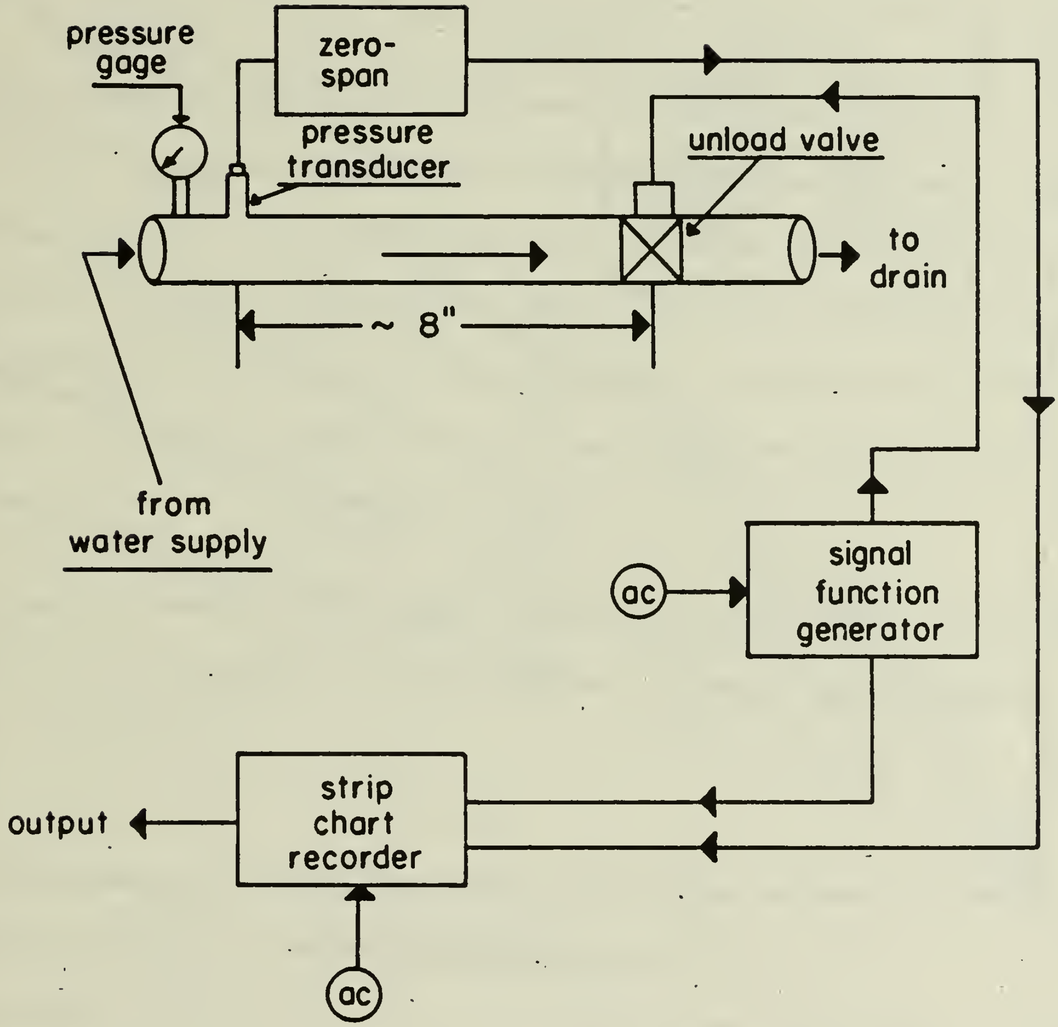

Figure 3. Valve Identification Apparatus

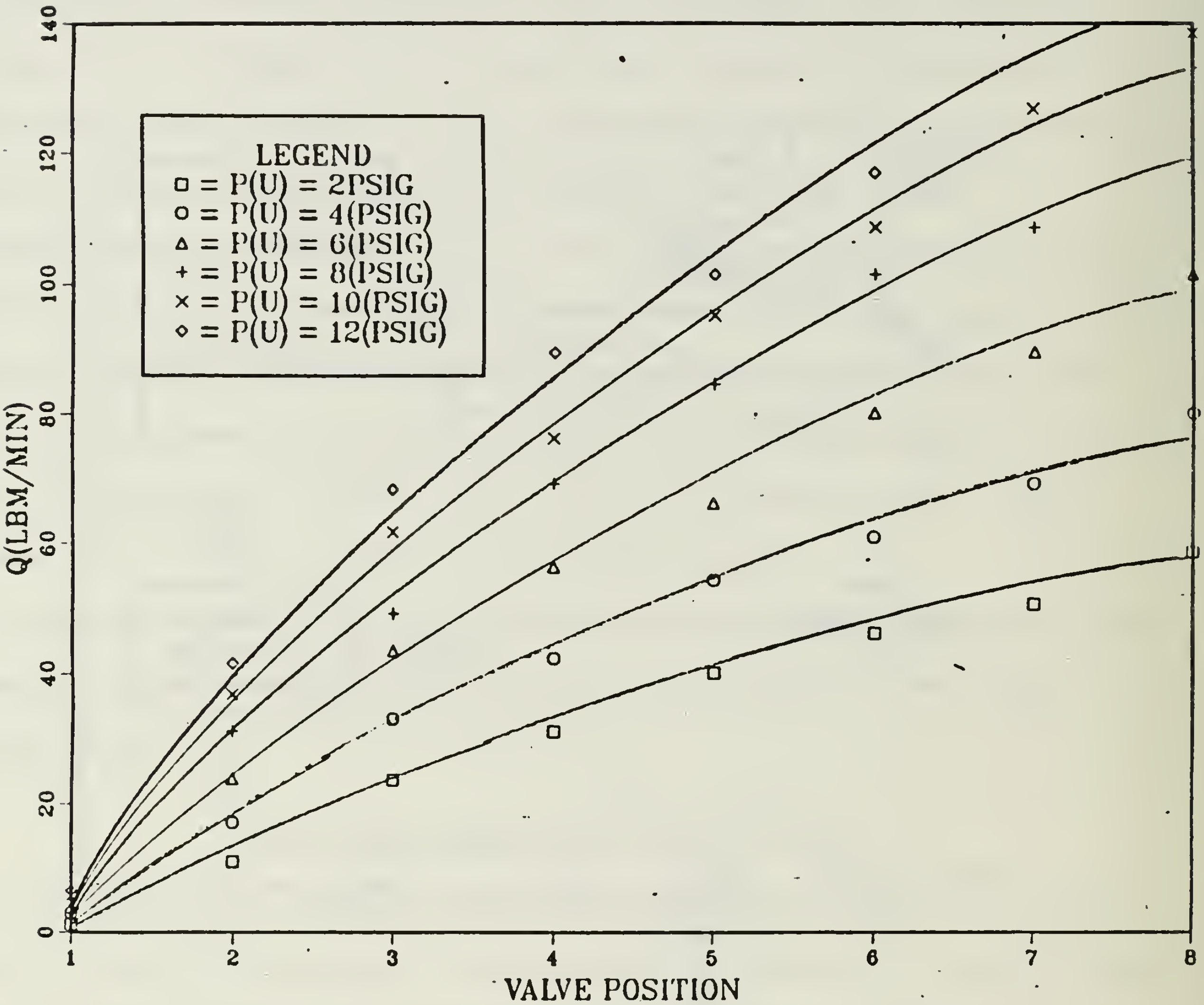

Figure 4. Steady-State Valve Performance

where q and v are perturbational values of the flow rate and the input voltage about a specified set operating point. Again, steady data can be used to characterize the valve orifice at some operating point, V_i;

$$q \cong \Delta Q = C_1 \Delta P)_{V_i = c}$$

where ΔP is the change in the upstream pressure, P_u.

To generate the dynamic data needed the equipment setup of figure 3 was used. Various operating point voltages, V_i, were selected and the flow rate manually adjusted by an inlet water supply valve to provide a specific value of P_u. A sinusoidal perturbational voltage was applied to the unload from the singal generator. The input voltage to the unload valve was described by:

$$V = V_i + v$$
$$V = V_i + A*\sin(\omega t)$$

The experiments generated small amplitude pressure variations, ΔP, about the mean level of P_u. Therefore, we calculated the perturbational flowrate from the valve relationship

$$q = C_1 \Delta P.$$

And, since the pertubational input amplitude v was known, we computed the transfer function magnitude, q/v, and generated Bode plots.

The experiments called for studying the effect of two parameters: A, the amplitude of the input sive wave and V_i, the input linearization point. For the experiments, an arbitrary valve of mean upsteam pressure was chosen at 10 psig. In the actual system the upstream pressure was to be 4 psig so we also investigated the effects of the mean upstream pressure variation on the valve transfer function.

III. Results

Figure 5 shows the effect of varying the amplitude, A, while holding the valve position, V_i=1.21 volts, and the upstream pressure, P_u=10 psig. This valve of V_i was chosen since it was near the steady state (V_i=0) and would permit closing as well as opening perturbations of the valve position. The plot shows that the midrange data (0.4 - 0.6 volts) converged. At amplitudes greater than 0.6 volts the upstream pressure variations became excessive and large changes in perturbational flow, q, were observed. For values smaller than 0.2 volts, excessively small amplitudes were seen in ΔP. Therefore, an amplitude of 0.4 volts was chosen for further work since it best represented the overall valve characteristic.

Figure 6 shows the effect of varying the mean valve position, V_i, while holding the amplitude, A=0.4 volts, and the upstream pressure, P_u = 10 psig, constant. From this plot we see a convergence of the data as the mean valve opening becomes greater. Also, the shape of the plots is consistent. The rolloff rates observed in figures 5 and 6 were found to represent a second order transfer function.

Figure 7 shows the effect of perturbations around various upstream pressures, P_u, while holding the valve position, V_i=1.21 volts, and the amplitude, A=0.4 volts, constant. It can be seen that the higher the value of P_u the more consistent the plots became, except for dispersion at the higher frequencies. In any case, the data was always clustured in a fairly narrow band.

This allowed us to specify the form of the transfer function of the unload valve as:

$$\frac{q(s)}{v(s)} = \frac{K}{\left(\frac{s}{\omega_1} + 1\right)\left(\frac{s}{\omega_2} + 1\right)}$$

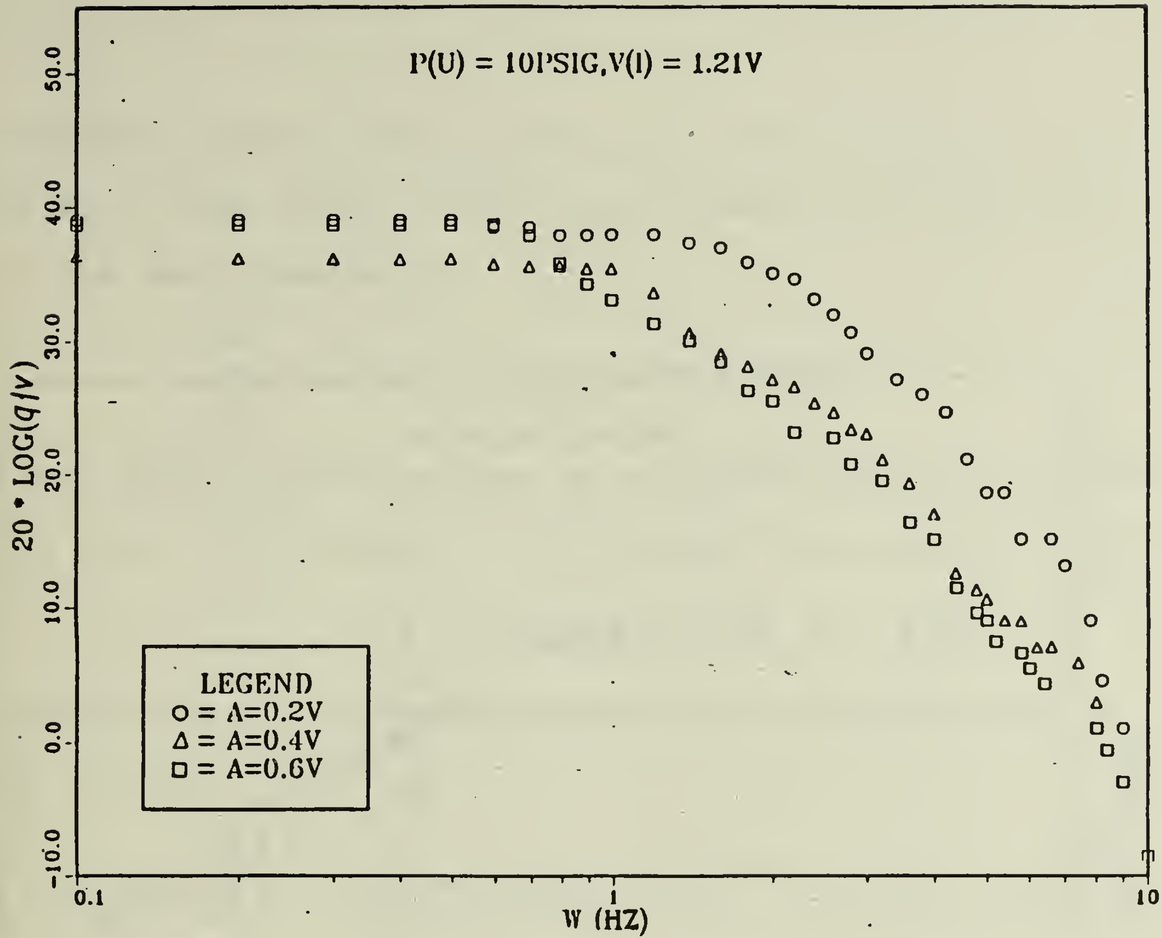

Figure 5. Bode Plot of Input Amplitudes

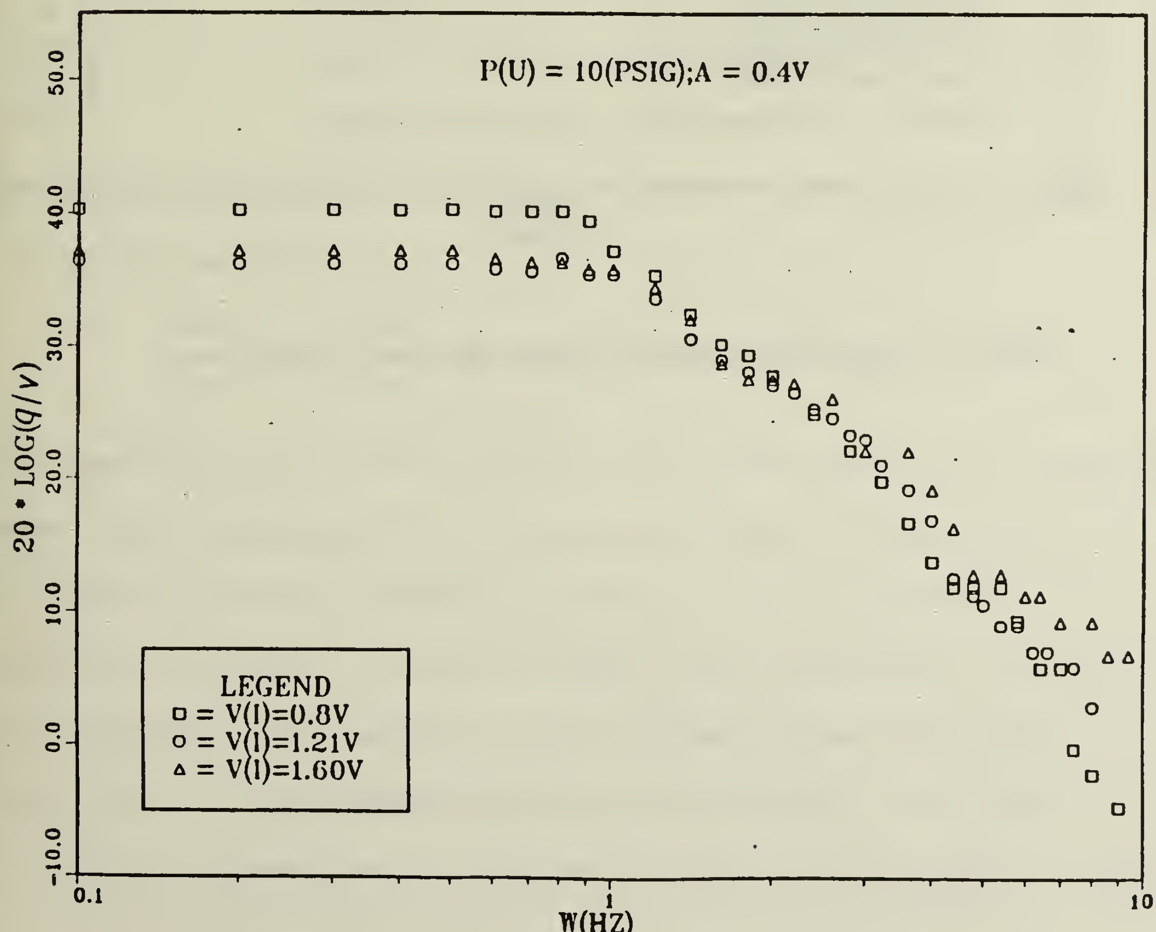

Figure 6. Bode Plot of Valve Mean Positions

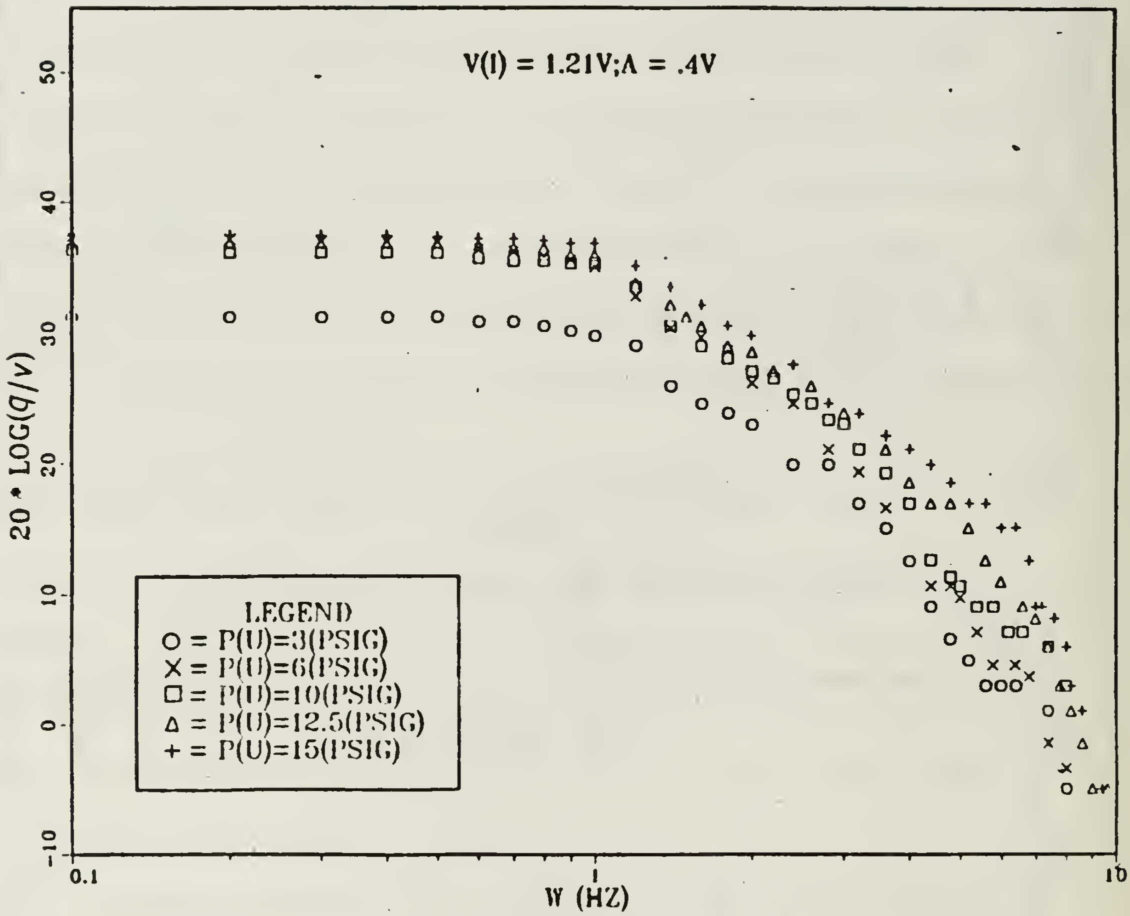

Figure 7. Bode Plots at Various Upstream Pressures

If we arbitrarily choose Q_i and V_i equal to zero, then the inverse laplace transform can be taken so that the time domain equation can be obtained. The resulting time domain equation is:

$$\frac{d^2q}{dt^2} + (\omega_1+\omega_2)\frac{dq}{dt} + (\omega_1\ \omega_2)q = K\ v\ \omega_1\omega_2$$

We next compare the above equation to the previous equation for Q from the steady state. If we multiply the valve steady-state sequation by $\omega_1\ \omega_2$ we get

$$(\omega_1\ \omega_2)\ Q = (\omega_1\ \omega_2)\ [-4.1704V_i^2 + 42.049V_i - 13.6472].$$

A nonlinear valve equation can be constructed by combining these equations to achieve:

$$\frac{d^2Q}{dt^2} + (\omega_1+\omega_2)\frac{dQ}{dt} + (\omega_1\ \omega_2)Q=(\omega_1\ \omega_2)[-4.1704V_i^2 + 42.049V_i - 13.6472].$$

From the Bode plot in figure 7, using the plot for P_u = 4 psig we get:

K=36.3977

ω_1=0.8Hz=5.027 radians/sec.

ω_2=1.7Hz=10.681 radians/sec.

Substituting these values into the general equation yields the final time domain nonlinear equation:

$$\frac{d^2Q}{dt^2} + 15.71\frac{dQ}{dt} + 53.69\ Q = -223.9V_i^2 + 2258V_i - 732.7$$

This equation for the unload valve was then incorporated into Johnson's CSMP system model and subjected to the same conditions that were used to generate figure 2. Figure 8 shows the results of this new model.

Comparison of figure 2 to figure 8 shows little improvement in the simulation response. This seeming failure has much to say about this modelling process. As discovered from the data of figure 7, the valve hardware had break frequencies of 5.03 and 10.68 Hz. These are both too high

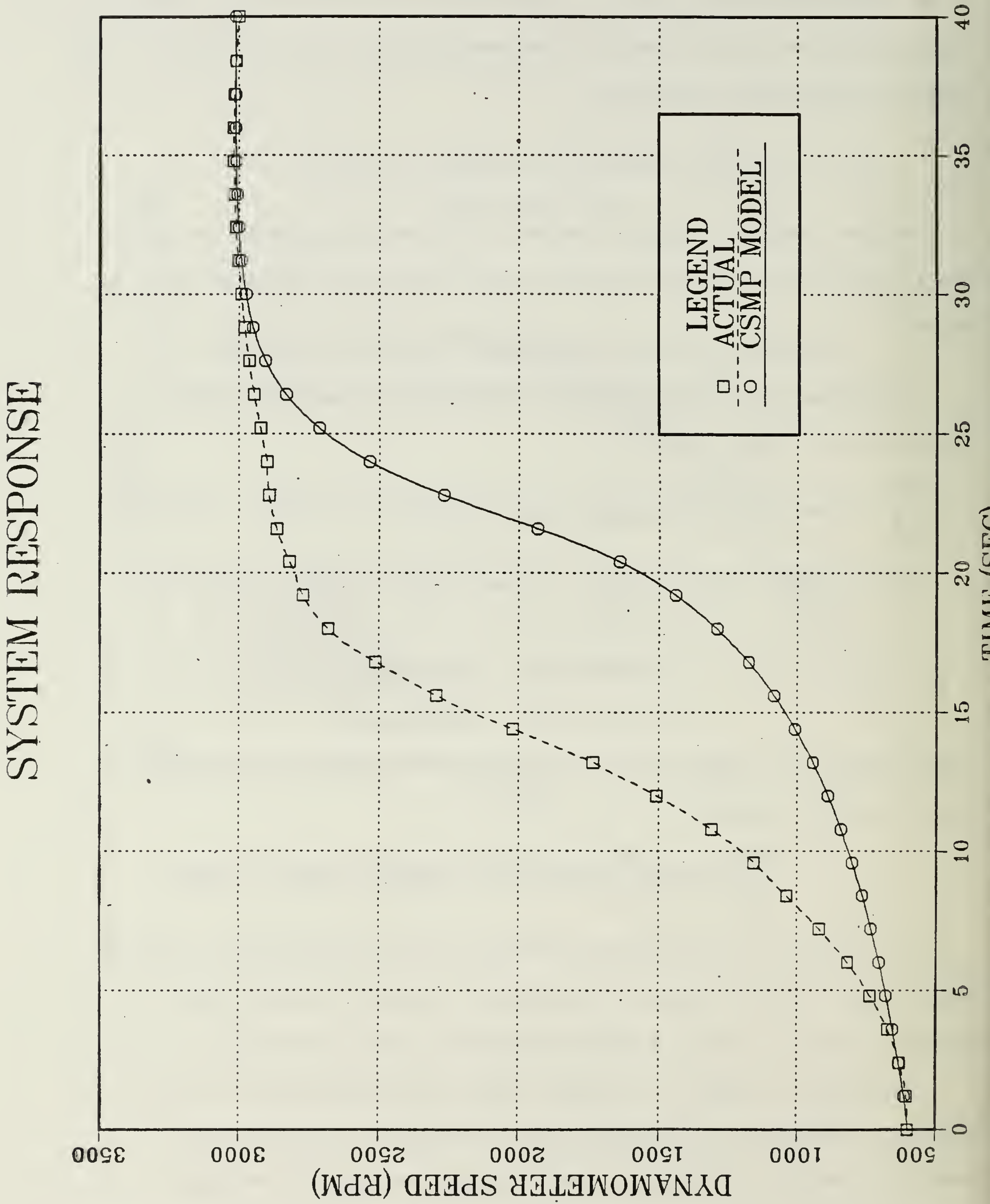

Figure 8. Dynamometer Unloading Response, New Model

to be of concern in the system response as shown in figure 8 (a break frequency of 0.1-0.2 Hz would be of more interest), so the continuing slowness of our simulated system is a logical result. Recall that in achieving this result we have assumed that a given valve pressure drop corresponds immediately with the steady-state flowrate. We also assumed that the fluid dynamics of the valve identification experiment were the same as those supplied by the dynamometer. Serious failure of the first assumption would mean that the valve would respond much slower than that now predicted. The failure of the second assumption is much more probable. For whatever reason, it is very likely that the steady-state ΔP vs Q relationship of figure 4 is somewhat too low in error. In order to correct for this, the gain of the valve was adjusted and improved results were obtained as shown in figure 9.

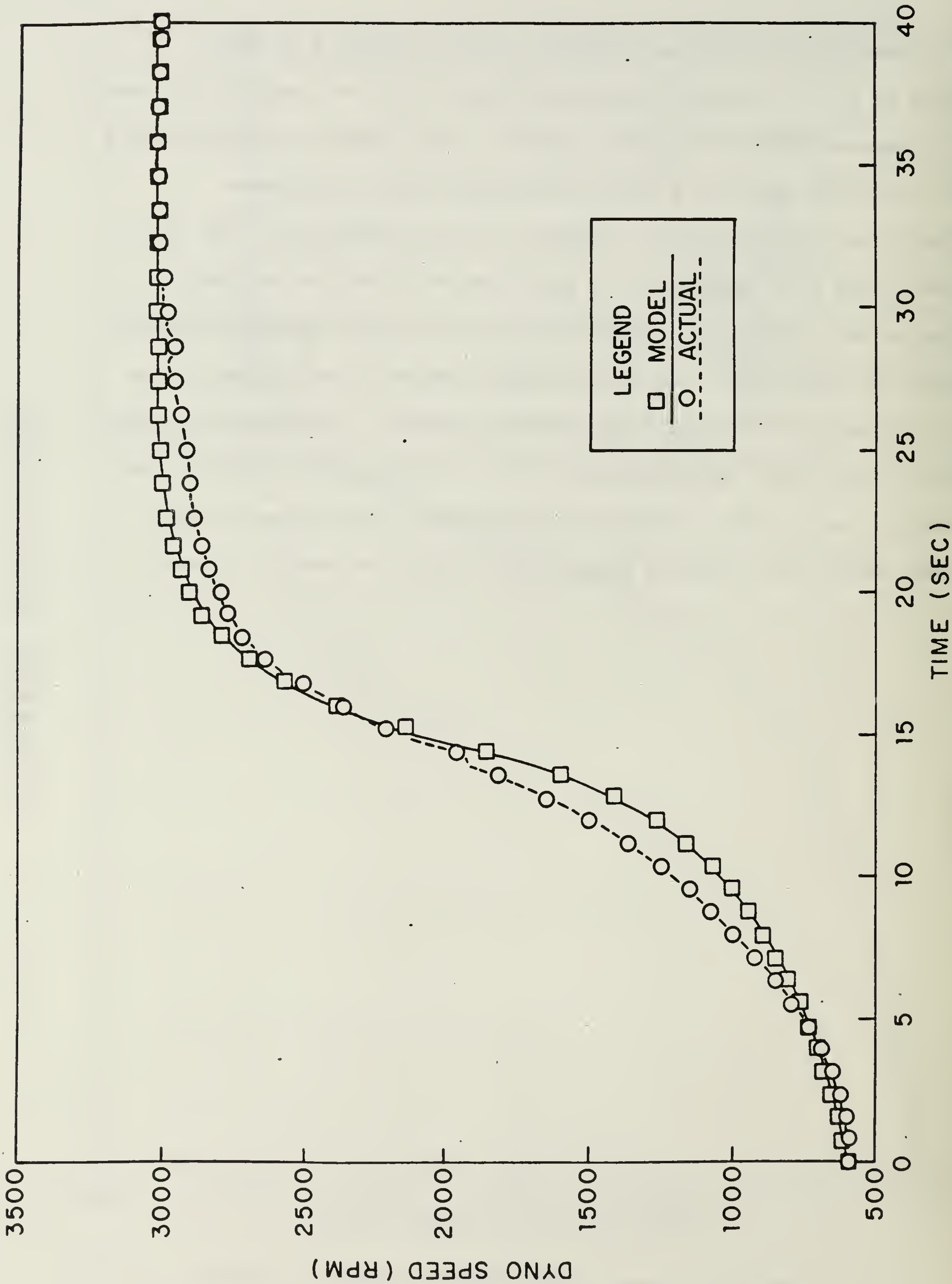

Figure 9. Dynamometer Unloading Response, Adjusted Gain

IV. Conclusions and Recommendations

The valve and dynamometer were too complicated to model in detail and some means of simplified modelling was deemed necessary. The present method of response measurement has the advantage of using performance data, with the difficulty of requiring data for the valve in a situation as close as possible to the real case. The interaction between the valve and the dynamometer somehow caused the valve to respond differently than it did without the dynamometer. This leads to an applicational difficulty: we have not characterized the valve for all systems, merely in this system alone. The model derived in this way works well for this application, but another application would probably require a fresh start. Thus, great care must be excersized when using old data from this approach for new design work, and even for modifying previous designs.

Since this method requires hardware and excitation of large frequency ranges, it may be destructive at or near resonant frequencies. Care must be excercized about resonances as well.

Variations in operating point variables did not seem to have a significant effect on either the model form or on parameter magnitudes. The experimental data clustered more or less along well defined regions. We have confidence that the second order nonlinear equation which describes the valve is accurate. While it turned out that the dynamics of the unload valve were not important in predicting system performance, we did not know this before we began. The present method gave us a means for determining the importance of the valve dynamics to the systems response.

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